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International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2012

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ABSTRACT

R410A is one of the main refrigerants used for air conditioning and heat pump systems in residential applications. It has zero ozone depletion potential but its global warming potential is about 2,000. In China and Japan, refrigerant R32 (GWP = 675, zero ODP) has been proposed, as possible replacement for R410A but this refrigerant is slightly flammable. HFO-1234yf is a refrigerant with low GWP (GWP = 4, zero ODP) that is currently being used in European car market as a possible replacement for R134a. The direct impact on the environment of these two refrigerants might be gauged based on their low GWP but a measure of their indirect contributions on greenhouse gasses effects is still an open question. Having a low GWP might not be sufficient to reduce the overall environmental impact if the energy performance of the heat pump system is significantly penalized. In addition, considerable amount of engineering and manufacturing work might be necessary for developing new components that are fit to work with these low GWP refrigerants.

This paper focuses on an experimental comparison of the drop-in energy performance and capacities of refrigerants R32 and R1234yf in a R410A heat pump split system for ducted HVAC in residential applications. The experiments were conducted in a large scale psychrometric chamber at Oklahoma State University and on a 5 ton heat pump unit that was commercially available. The experiments were conducted for cooling and heating modes of the unit and the outdoor temperature was varied from 17°F (-8°C) to 115°F (46°C). Cooling tests at AHRI standard rating conditions were performed and the refrigerant charge was optimized. Two additional conditions were considered with high outdoor temperatures of 110°F (43°C) and 115°F (46°C) to analyze the condenser pressure and discharge temperature of the refrigerant when the unit run at extreme high ambient off-design conditions. The paper highlights some challenges related to R32 and R1234yf and discusses an optimization of the thermal expansion valve when these refrigerants are directly retrofitted in the unit. It was observed that the COP was higher for both R32 and R1234yf as compared to R410A with some peculiar behavior at very high temperatures, for which R32 COP was lower than that for R410A. The capacity for refrigerant R1234yf was 40 to 50% lower and the management of the charge as well as the adjustments of the TXV was problematic without any additional modification to the system.

1. INTRODUCTION

The refrigeration and air-conditioning industry has experienced several changes since the last two decades in order to meet the terms of the Montreal Protocol. A gradual phase-out of refrigerants that deplete the ozone layer has occurred and CFCs to HCFCs have been retrofitted with more environmentally friendly refrigerants. However, HFCs and new blends used in modern air conditioning and heat pump systems have still global warming potential (GPW) that is of concern. For example R134a is a hydrofluorocarbon refrigerant with ODP of zero but with GWP of about 1,430 (Forster *et al.,* 2007) and the European Union directed the phase out of R134a in automobile air conditioning in the coming decade.

Refrigerant R410A has good thermal and transport properties and high volumetric capacity for AC applications, but it might have unfavorable effects on the environment in case of leakage. Energy conservation and environmental concerns are the core reasons for the tremendous growth of high performance and environmentally friendly heat transfer fluids. Research efforts are ongoing in the HVAC and Refrigeration industry that explore the performance of refrigerants that have low global warming potential, referred to as LGWP refrigerants, when used for R410A retrofit applications. This paper presents data of the thermal performance of two Low Global Warming Potential (LGWP) refrigerants, that is, refrigerant R32 (GWP ~ 675) and R1234yf (GWP ~ 4). The experimental study was conducted on a 5 ton nominal capacity heat pump ducted split system at design and off-design conditions with extreme high temperatures. The experiments were conducted in a large scale psychrometric chamber at Oklahoma State University and refrigerant cycle temperatures and pressures were directly measured. The thermal and transport characteristics of the refrigerants are discussed for straight drop-in tests, in which none of the original components of the unit were modified, and for soft-optimization tests, in which the original R410A thermal expansion valve was modified for further optimization of the refrigeration cycles for the case with refrigerants R32 and R1234yf.

2. REFRIGERANTS CHARACTERISTICS

Refrigerant R410A is a near-azeotropic blend of R32 and R125, with a critical temperature of 72.8°C and a critical pressure of 4.86MPa. It does not contribute to the ozone depletion and it has been adopted in air conditioning and heat pump systems for residential and light commercial applications. R410A has a high volumetric cooling capacity, which means that this refrigerant can absorb significant amount of heat from the air in direct expansion evaporators for a unit volume of refrigerant. R410A operates at higher pressures than R22 and its GWP is 2,088, which is higher to that of R22 (Forster *et al.*, 2007). In case of leakage or improper refrigerant charge management, the direct contribution of R410A to the greenhouse effect might be assessed by considering its GWP. The system coefficient of performance (COP), and system reliability and life cycle time must be also considered in order to evaluate the refrigerant indirect contributions to the greenhouse effects. The indirect contributions are linked to the carbon emissions due to the power consumption of the unit during service and due to the energy and materials required for building the system (Minor and Spatz, 2008). In this work, when comparing the performance of other refrigerants with respect to R410A the authors purposely choose to use the same system. One could argue that if the same system is adopted for all three refrigerants, then the energy and material required to develop the system are also the same and thus by comparing the energy performance and capacity of the unit the indirect contributions can be estimated. In the present work the authors took this approach and the data were normalized with respect to that for R410A. It should be noted that the heat pump unit was developed for R410A and it was not necessarily optimized for the other two refrigerants, that is, neither for R32 nor for R1234yf. Straight drop-in performance tests in this work served as starting point for further research on the equipment that can be designed ad-hoc for these specific refrigerants. However, modifications of the thermal expansion valves were considered minor changes to the systems that did not involve any additional material or significant extra design and cost efforts. Thus, additional tests were carried out by performing an optimization of the expansion valve for R32 and R1234yf. The optimization procedure was also conducted for R410A, to guarantee that the R410A performance data were representative of the maximum COP and capacity of the unit at different operating conditions. The measured performance data of R410A were chosen as baseline for the COPs and capacities reported in this paper.

2.1 Refrigerant HFO R1234yf

Refrigerant R1234yf is a hydrofluoroolefin 2,3,3,3-tetrafluoroprop-1-ene that falls in the category of partially fluorinated olefins (Minor and Spatz, 2008). R1234yf has a GWP of 4 and is low in toxicity. It is classified as a low flammable (2L) refrigerant and has been proposed to replace R134a in automobile air conditioning systems. Its critical temperature is 94.7°C and the critical pressure is 3.38 MPa. While its vapor pressure is similar to that for R134a, its thermodynamic properties do not quite match those for R410A (Reasor *et al*, 2010). The ratio of operating pressure when compared to R410A is 40% lower (Endoh *et al*., 2010). The tests results for the compatibility of R1234yf and lubricant at typical operating conditions seems to suggest that the lubricating properties are unhampered by the use of R1234yf (Yana Motta *et al.*, 2010). Some modeling work in the literature showed that when R1234yf was retrofitted in R410A AC systems, the COP was similar to that for R410A but the cooling capacity generally decreased by as much as 50% (Leck, 2010). It is one objective of the present work to verify these findings for the case of a 5 ton split heat pump system for ducted residential applications. Experimental data were taken by using identical equipment and by using the same unit with an optimized expansion valve for R1234yf.

2.2 Refrigerant R32

Refrigerant R32 is a difluoromethane and it is a pure fluid with a GWP of 675. The critical temperature for R32 is 78.2°C and its critical pressure is 5.8 MPa, which are close to that for R410A. R32 belongs to low flammable refrigerants class 2L but might pose a greater fire hazard than R1234yf as a result of its faster flame propagation speed. R32 has a higher volumetric capacity than R410A and thus it has the potential of higher COPs. However, R32 yields to generally higher compressor discharge temperatures. This characteristic seems to adversely impact the system reliability because of metal fatigue of the valves and thermal stress of the lubricant (Leck, 2010). Taira *et al*. (2011) observed that since R32 has a larger volumetric capacity than R410A the amount of R32 charge can be reduced by 50% of that of R410A for same system performance. Lower refrigerant inventory might decrease the direct impact to the greenhouse gases effects in case of refrigerant leakage. R32 has been proposed as possible replacement for R410A in various countries, especially in Asia, but its flammability characteristics still pose some serious concerns when it is considered for residential applications. To mitigate the negative traits of R32 Leck (Leck, 2010) suggested using R32 in refrigerant blends.

3. EXPERIMENTAL SETUP, TEST CONDITIONS AND PROCEDURES

The experimental campaign was carried out in a large-scale climate control psychrometric chamber at Oklahoma State University (OSU). The new psychrometric facility is shown in Figure 1(a) and it consisted of two adjacent air conditioned rooms that were controlled over a wide range of conditions with and without a live load in it; one room artificially reproduced the outdoor climate conditions while the other room was employed to simulate the indoor environments and replicated indoor comfort conditions with thermal loads up to 20 tons of refrigeration (70 kW). Cremaschi and Lee (2008) provided details about the facility design, construction, and specifications for the psychrometric chamber used in this work. The facility was fairly large with respect to the equipment tested in this work in order to guarantee uniform temperatures and humidity around the equipment and to minimize thermal and fluid-dynamic interferences between the air streams to/from the unit with the walls and ceiling of the rooms. Figure 1(a) shows the layout of the heat pump split system ducted inside the psychrometric facility during the laboratory measurements. A condensing unit, which consists of an outdoor coil, fan, and a compressor, was installed in the outdoor room while the indoor coil-blower assembly was ducted inside the indoor room. Two refrigerant pipelines connected the indoor assembly to the outdoor condensing unit. Figure 1(b) shows photos of the outdoor condensing unit and indoor blower-fan assembly used in the present work. Two sets of dry bulb and wet bulb probes (ASHRAE standard 116) were installed in front of the condensing unit and of the indoor coil. A third probe was installed downstream the coil to measure the humidity ratio of the air stream leaving the indoor coil. A grid of five thermocouples was installed inside a 0.3 m by 0.6 m (1 ft. by 2 ft.) rectangular duct, which was mounted at the top of the indoor coil assembly (see Figure 1(b) photo on the left side). Special care was taken when insulating the indoor unit assembly in order to minimize the heat losses to the ambient. Flow nozzles were used to measure the air flow rate according to the ASHRAE standard 41.2. Pressure taps were also installed at the outlet of the indoor blower-coil assembly according to the specifications of the ASHRAE standard 41.2. During the tests, the dry bulb temperature was controlled within ±0.1°C and the wet bulb temperature was controlled to within ±0.2°C with respect to the set points. More details about the chamber in-house instrumentation and controls can be found in Worthington *et al* (2011).

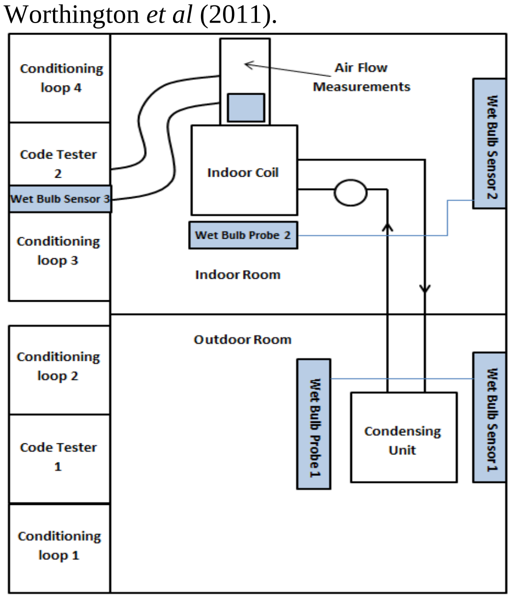

Figure 1(a): Floor plan of heat pump split system inside OSU large-scale climate control psychrometric chamber

Figure 1(b): Photos of the indoor blower/coil assembly (left side) and outdoor condensing unit (right side).

3.1 Experimental setup

The heat pump used for the experiments was a R410A heat pump split system commercially available off-the-shelf and with a rated capacity of 5 ton (17.6 kW). The unit was powered on 208VAC, 3-phase, 60Hz current in typical ducted HVAC residential applications in U.S. The unit had an A-shape fin-and-tube direct expansion evaporator with constant speed blower and a tube-and-fin condenser with constant speed fan. An hermetic reciprocating refrigeration compressor was in the condensing unit and it had 47.28 cm^3/rev (2.885 in^3/rev) displacement volume and run at 3450 RPM at 60 Hz. The compressor was pre-charged by the manufacturer with 1,567 milliliter (53 oz) of POE lubricant, which had a viscosity of 30 cSt at 40°C (104°F). A suction accumulator was installed by the manufacturer. The unit had two TXVs and a 4-way reversing valve: one TXV was installed at the inlet of the indoor coil for cooling mode operation while the second TXV was at the outdoor coil for heat pump operation. The system was installed inside the psychrometric chamber as shown in Figure 1(a) and the suction horizontal line was about 12.5 meters (41 feet) long from the indoor assembly to the outdoor condensing unit and it had an internal diameter of about 2.6 cm (1.02 inch) (1 inch nominal copper pipe type L). The unit did not have a discharge oil separator and additional 650 milliliters (22 oz) of POE lubricant were charged into the compressor suction port to compensate for the oil that could be retained along the suction line during the unit operation. The unit was instrumented as shown in Figure 2. Temperature and pressure sensors were installed at the inlet and outlet of each component of the refrigeration system. The discharge pressure and temperature (P_1 and T_1 in Figure 2) were located on the refrigerant discharge line after the 4-way. The distance between these sensors and the compressor discharge port was about 0.6 m (2 ft) of pipeline. The 4-way valve and the refrigerant pipelines were well insulated to prevent heat losses to the ambient. However, some heat exchange was expected to occur between the hot vapor in the discharge line and the cold vapor in the suction line when the refrigerant crossed the 4-way valve. valvA Coriolis type flow meter was also mounted in the refrigerant liquid line. Several surface thermocouples (indicated as Ts in Figure 2) were positioned along the refrigeration system to estimate the heat losses in the pipelines and in the 4-way reversing valve. Several surface thermocouples were also installed on the compressor crankcase to monitor the compressor housing temperature. A wattmeter was used to measure the electric power of the unit and the unit capacity was measured from the air side (primary method) and from the refrigerant side (secondary method). The range and accuracy of the instrumentation is given in Table 1 below. The uncertainty of the air side capacity and COP were estimated to be within 3% and 4%, respectively. During the optimization of the expansion valve, the set up was modified as shown in the bottom box of Figure 2 referred to as “modified system option”. The thermal expansion valve was replaced with a manual expansion valve, an additional pressure transducer, and an in-line temperature sensor (P_4 and T_4 in Figure 2), which were installed at the inlet of the distributor of the direct-expansion evaporator.

Table 1: Instrumentation used in the test apparatus

Sensor	Type	Range	Accuracy
Absolute refrigerant pressure transducer	Piezoelectric	0 – 3,550 kPa (0 – 500 psig)	±0.25% F.S.
Differential air pressure transducer (for air flow rate measurments)	Piezoelectric	0 – 0.75 kPa (0 – 3 in wc.)	±0.14% F.S.
Mass flow meter	Coriolis	45 – 454 kg/hr (100 – 1000 lb_m/hr)	±0.1% F.S
Thermocouple	T-type	17 to 115 °F (-8 to 46°C)	±0.5°F (0.3°C)
RDTs	100 Ohm DIN Platinum RTDs with 1/3 DIN Accuracy	17 to 115 °F (-8 to 46°C)	±0.2°F (0.1°C)*
Unit power	Watt transducer with current transformer	0- 50amps 0-230 VAC, 3 phase	±0.2%
Refrigerant charge	Electronic Scale	30 kg (66 lb_m)	±0.1 kg (0.2 lb_m)

*in-house and in-situ calibration with high precision temperature bath

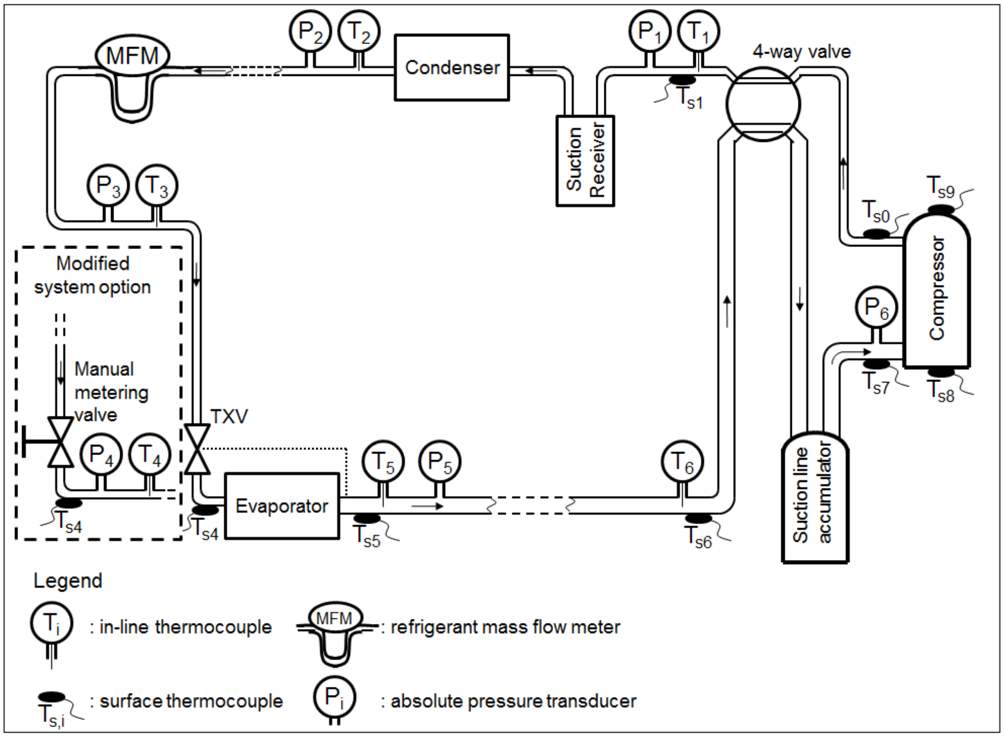

Figure 2: Schematic of the heat pump system and layout of the instrumentation

3.2 Test procedures during straight drop-in tests

Improper refrigerant charge can reduce the efficiency of the system by 10 to 20 percent in the field applications (Downey and Proctor, 2002). Both undercharge and overcharge of refrigerant in the system could affect the lifespan of the equipment, capacity and efficiency. Undercharge decreases the capacity and COP because of the drop in the refrigerant mass flow rate. The flow rate decreases mainly because of a decrease in the evaporating pressure and an increase in suction superheat. Overcharge tends to increase the capacity but the energy efficiency of the system decreases (Kim and Braun, 2010). In this work, first the unit was tested with R410A and the refrigerant charge was optimized for maximum COP at AHRI 210 A rating test conditions (AHRI, 2010). In each test, once the control tolerances were satisfied and steady state conditions were achieved, data were recorded for 1 hour with a sample rate of 2 seconds and the average COPs and cooling capacities were calculated. Then the refrigerant charge was increased in steps of 0.23 kg (0.5 lb_m) at a time. Each test was conducted multiple times before moving to the next one to guarantee that the measurements were repeatable. The refrigerant charge that provided the maximum COP was selected as to the optimum charge in the system. During the charge optimization process the degree of vapor superheat at the compressor suction was above of at least 2.2°C (4°F). This limit guaranteed safe operation of the compressor. Once the optimum refrigerant charge in the system was determined, the unit was run for a broad range of temperatures from -8.3 °C (17°F) to 46°C (115°F) and in both heating and cooling modes at full load conditions. Additional tests were conducted at extreme high temperature conditions of 43° and 46°C (110° and 115°F) to investigate the differences and similarities (if any) of the refrigerant condensation pressures and compressor discharge temperatures of R32 and R1234yf with respect to those of R410A. These ambient conditions were extreme but often occur during the summer months in the South and Midwest regions of the United States, as well as in the Middle East areas and Southeast Asia.

Once the tests of R410A were completed, this refrigerant was slowly recovered in 2 to 3 hours. Then the system was vacuumed for about 30 to 40 minutes and R32 was charged into the system. R32 charge optimization was conducted first at A test cooling conditions until the optimum charge of R32 was identified. Then the system with optimum charge was run at other outdoor temperatures and in both cooling and heating modes. A similar procedure was adopted for R1234yf; once the optimum charge for R1234yf was determined at AHRI A test cooling conditions it was modified during the remaining tests at higher and lower temperatures only if the degree of superheat became lower than 2.2°C (4°F). At the end of the experiments with R1234yf, R410A was recharged into the system and the performance tests were repeated. These tests served to verify the hypothesis that any potential removals of compressor lubricant from the system during the recovery of the previous refrigerant did not produce any measurable effects on the COPs and capacities of the system. In other words, since the first and last series of tests with R410A showed similar data and R32 and R1234yf were tested in between those two series, the results presented in the next sections were considered independent from the chronological order on which the refrigerants were charged and tested in the system.

3.3 Test procedures during the soft optimization of the expansion valve

The unit TXV controls the degree of superheat at the evaporator outlet based on R410A cycle characteristics. Once the straight drop-in tests with R32 and R1234yf were completed, the original TXV of the unit was replaced with a manual expansion valve in order to set different degree of superheat and pressure ratios for R32 and R1234yf. For each refrigerant, an initial mass was charged into the system close to the one of the straight drop-in tests. Then the opening of the expansion valve was varied in search of further improvements of the COP. If similar COPs were measured, the expansion valve opening was optimized for augmenting the capacity of the system. These tests were carried out with the unit in cooling mode only. The refrigerant charge was varied in incremental steps of about 0.23 kg (0.5 lb_m) and for each charge the expansion valve opening was varied in a parametric fashion. The optimization procedure of the expansion valve opening and refrigerant charge continued until either a maximum COP was identified or the degree of superheat at the compressor suction became lower than 2°C (4°F).

4. RESULTS AND DISCUSSION

4.1 Experimental validation of the COP and capacity measurements

In order to demonstrate the accuracy of the COP and capacity measurements, two types of experimental validation of the test apparatus were carried out. The unit cooling performances with R410A were compared with the unit cooling rating performance data provided by the manufacturer. The measured energy efficiency ratios (EER) were within 3% of the manufacturer data. Good confidence for the accuracy and repeatability of the measurements for air side capacity was established by comparing measured air flow rates, supply dry bulb and wet bulb temperatures, and external static pressure across the indoor coil with the corresponding data provided by the manufacturer. A second experimental validation was the heat balance between the air side and refrigerant side of the indoor coil during the actual tests. The heat balance for all cooling tests in this work was within an average of 2.8% and in one instance it increased to 3.9% due to the lack of enough refrigerant subcooling at the condenser outlet for one test with R1234yf. In such case two phase refrigerant circulates inside the flow meter and the authors speculated that the error in the refrigerant mass flow rate measurement was responsible for this increased heat balance error.

4.2 Charge Optimization Results

Figure 3 shows the data of the charge optimization tests for refrigerant R32. The COPs and capacities were normalized with respect to that for optimum charge of R410A. The compressor pressure ratios, Pr, are shown on the x-axis and they were also normalized with respect to R410A. For example, for 5.9 kg of R32 charged in the unit there are two solid diamond data points in Figure 3 that corresponds to two positions of the expansion valve. The COPs of the unit with this charge of R32 were about 4% higher than that for R410A and the corresponding cooling capacities were about 8% higher than R410A. The pressure ratios with R32 were lower than that for R410A. When the refrigerant charge was increased to 6.6 kg and the expansion valve was properly adjusted, the COP for R32 was 5.5% higher than that for R410A. A further increase in charge slightly decreased the COP until the compressor superheat reached the lower limit. The data of Figure 3 indicate that when the unit was retrofitted with R32 and the expansion valve was properly adjusted, the COP was up to 5.5% higher, cooling capacity was up to 10% higher, and the cycle pressure ratio was about 0.98 times lower than that for R410A.

Figure 4 shows the charge optimization for R1234yf. In this case, the optimal charge of R1234yf was 9.5 kg and the COP was 1% higher than that for R410A. The corresponding capacity of the unit with R1234yf was 46% lower than that for R410A and the R1234yf pressure ratio was 0.8 times lower than that for R410A. From the data of Figure 4 it was evident that even when R1234yf provided similar COPs, the unit cooling capacity was significantly reduced and a soft optimization of the expansion valve was not sufficient for augmenting the cooling capacity of the unit when R1234yf was used. From this perspective the authors concluded that R1234yf was not a suitable direct drop-in replacement for R410A and some other modifications of the system would be required.

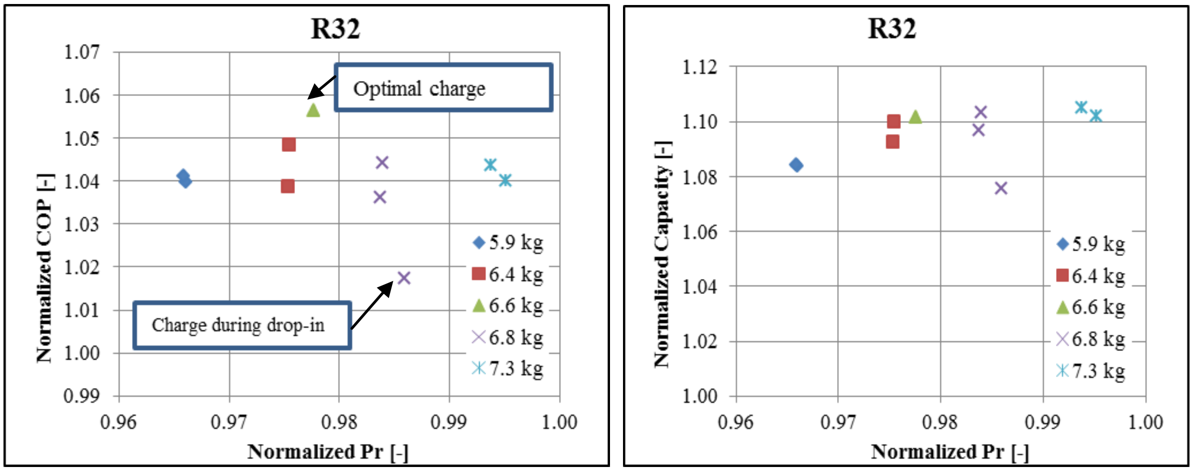

Figure 3: Results from the R32 charge optimization.

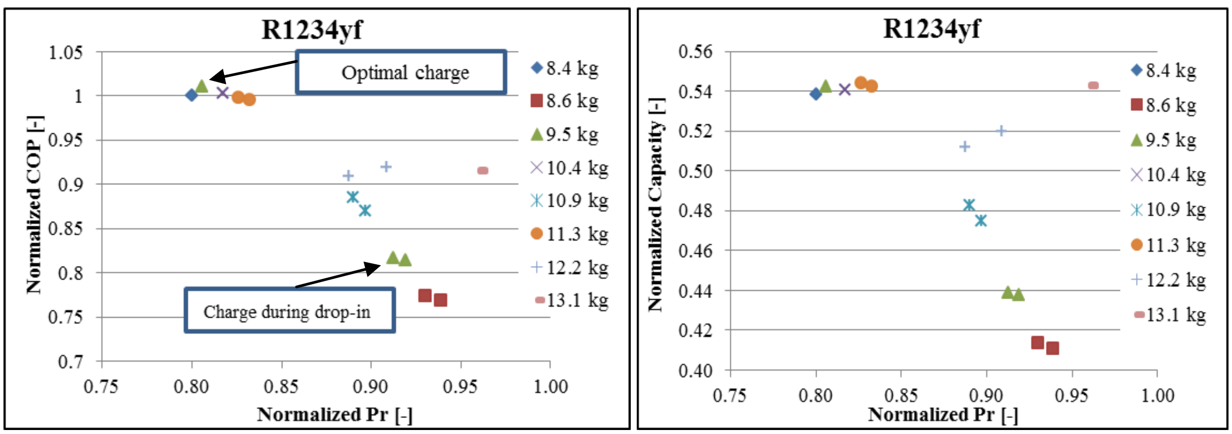

Figure 4: Results of the R1234yf charge optimization.

4.2 Drop-in Performance Tests

The drop-in performances of the heat pump with R32 and R1234yf are shown in Figure 5 for both cooling and heating modes. The COPs are normalized with respect to that for R410A at similar outdoor temperatures operating conditions. While in cooling mode R32 had similar COPs as R410A, a significant increase of COP was observed in heating mode from 10 and 17%. The COP for R1234yf was higher in both cooling and heating modes with an exception at B-test where the COP of R1234yf was 10% lower than that of R410A. This was due to the fact that the TXV used for drop-in tests was originally designed for R410A and it was operating at off-design conditions for R1234yf at B conditions. It was also noted that the degree superheat for the B test was quite high, almost 10.5°C (19.3°F). This higher degree of superheat penalized the compressor performance and thus decreased the system COP for R1234yf in the case of AHRI B-test cooling conditions.

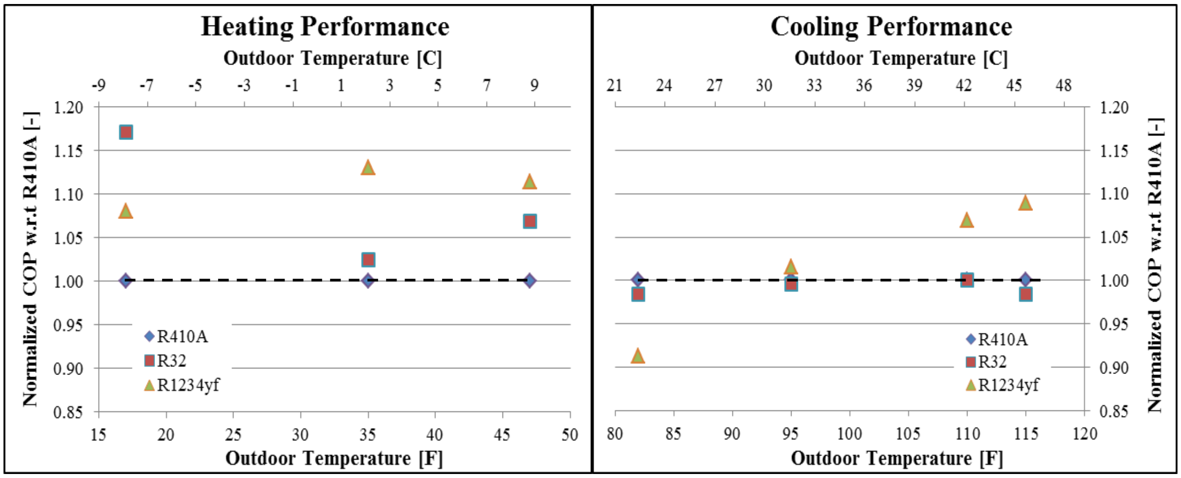

Figure 5: Normalized COP with respect to R410A for R32 and R1234yf.

The capacity for R32 cooling mode was 10% higher than R410A, especially at extreme high temperatures as shown in the right plot of Figure 6. In heating mode, the capacity for R32 was comparable to R410A. The capacity of R1234yf was about 50% lower for the entire range of outdoor temperatures and in both cooling and heating modes. It was also observed that the 1.13 kg (2.5 lb_m) of R1234yf had to be taken out from the unit during the extreme high temperature tests in order to achieve sufficient degree of superheat of at least 2°C (4°F). Thus, the charge management of R1234yf might be a challenge if R1234yf is retrofitted in R410A heat pump units for residential applications without any modification of the system refrigerant receiver.

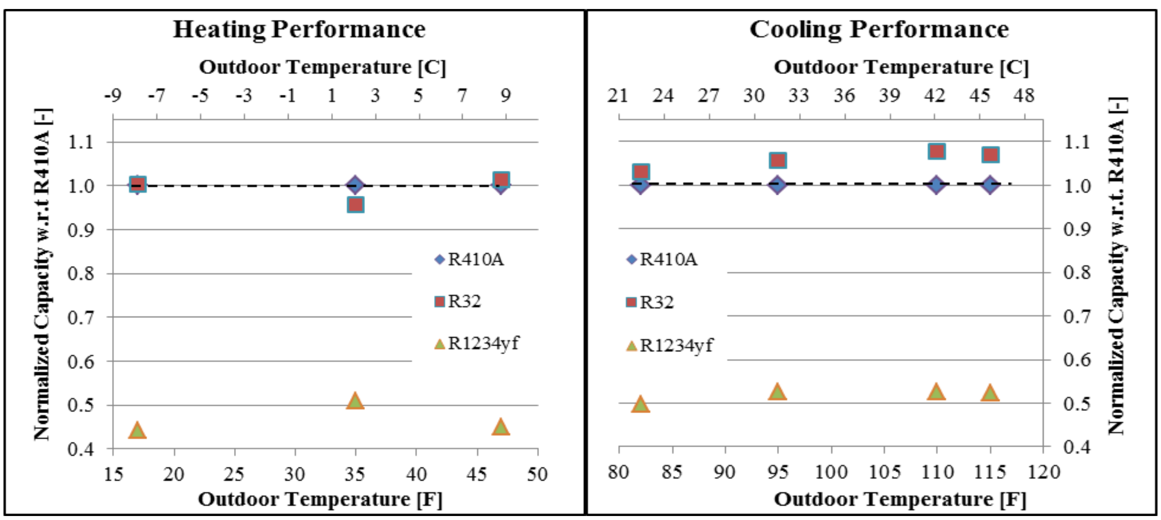

Figure 6: Normalized capacity with respect to R410A.

The power consumed by the compressor for R1234yf was about 48 to 60% lower than that for R410A. This was due to lower refrigerant flow rates and lower pressure ratios of R1234yf. With R32 the compressor power was 7 to 9 % higher than the baseline with R410A. The compressor performances are discussed by analyzing the difference of suction vapor specific volume of the refrigerants and the thermal efficiencies of the compression process during the actual experiments. These data are shown in Figure 7 for various outdoor temperatures. During the unit run with the original TXV, the suction specific volumes for R1234yf measured from the pressure and temperature data taken at the compressor suction port, were about 45 to 60% higher than R410A. R32 specific volumes were about 35 to 40% higher than R410A. The thermal efficiency of the compressor, normalized with respect to R410A, was generally higher for R32 and lower for R1234yf. It was also observed that the discharge temperatures for R32 was higher than for R410A, especially for the extreme high temperature tests as shown in the right plot of Figure 8. The discharge temperature of the compressor with R32 was about 20°C (37.2°F) higher that R410A at AHRI A cooling conditions and up 30°C (54.6°C) at extreme high temperatures. For R1234yf the discharge temperature was always lower than R410A and the data in Figure 8 show the difference in discharge temperature of R1234yf and R32 with respect to R410A for a broad range of outdoor temperatures.

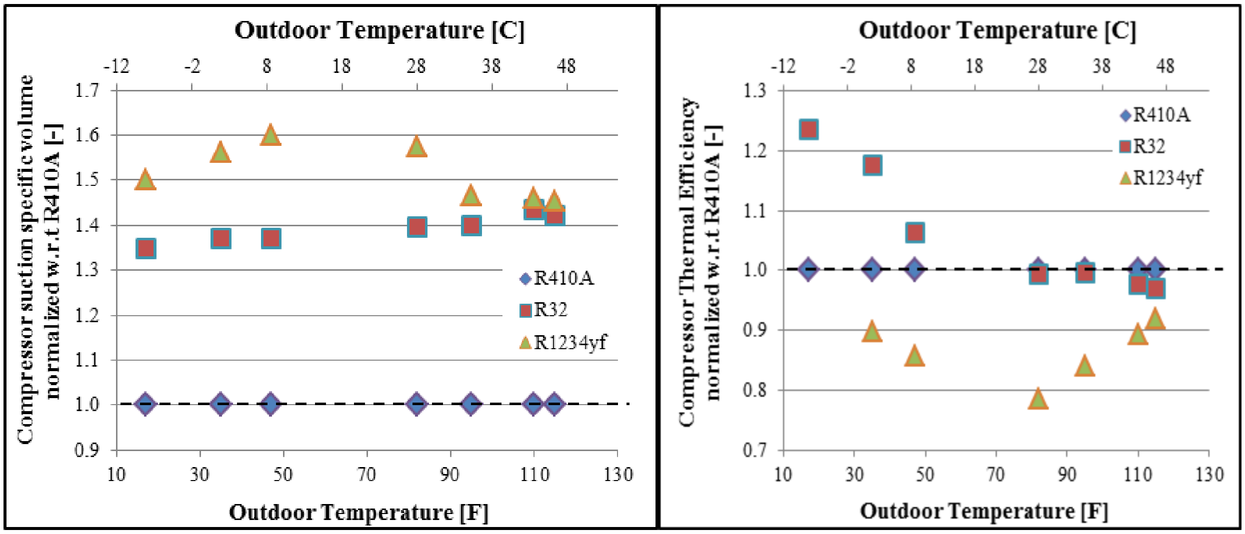

Figure 7: Compressor suction specific volume and thermal efficiency for R410A (baseline), R32, and R1234yf

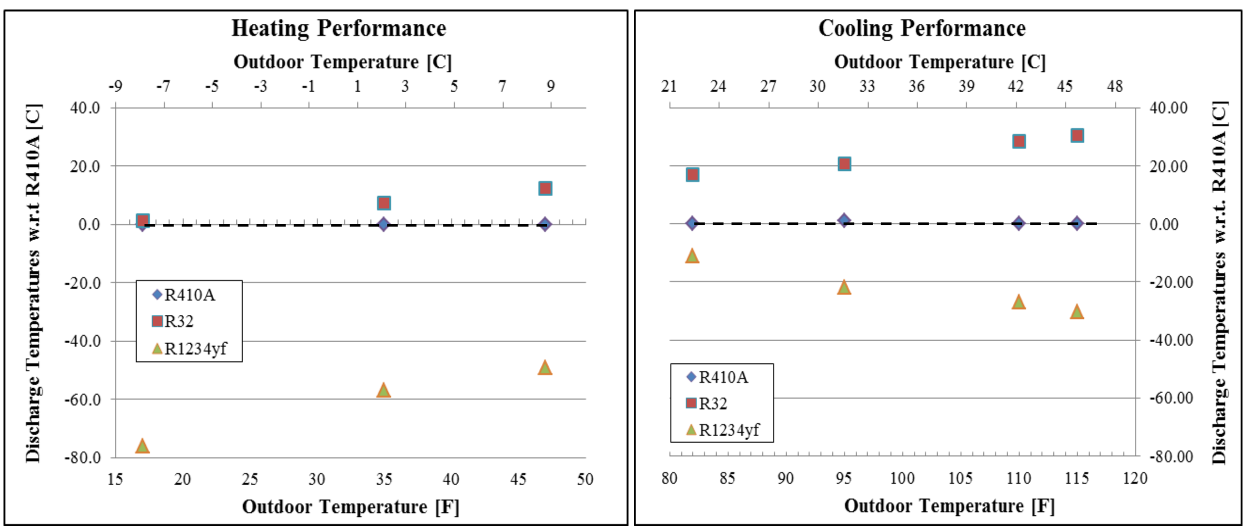

Figure 8: Compressor discharge temperatures for R410A (baseline), R32, and R1234yf

5. CONCLUSIONS

This paper presents data of drop-in energy performance and capacities of two low GWP refrigerants in a R410A heat pump split system for ducted HVAC residential applications. The experiments were conducted for cooling and heating mode of the unit and the outdoor temperature was varied from 17°F (-8°C) to 115°F (46°C). Cooling tests at AHRI standard rating conditions were performed and the refrigerant charge was optimized. Two additional off-design conditions were considered with outdoor temperatures of 110°F (43°C) and 115°F (46°C) to analyze the behavior of the refrigerant at extreme high temperature ambient. The findings from this work suggested than for refrigerant drop-in applications, R32 has comparable heating and cooling capacities as those for R410A and also similar COPs. The discharge pressures and discharge temperatures were higher than those for R410A, especially for moderate to extreme high temperature conditions. Too high discharge temperature and pressure of R32 in extreme high temperature conditions was a concern for the safe operation of the unit and might be a concern for the compressor lifetime cycle. Refrigerant R1234yf provided similar COPs as R410A but this refrigerant had rather low capacities with respect to those for R410A. Optimizing the expansion valve improved the R1234yf capacity by up to 10% with respect to drop-in capacities but it was still 46% lower than that for R410A at similar operating conditions. From this point of view R1234yf was not a drop-in replacement for refrigerant R410A when considering a 5 ton commercially available heat pump split system for ducted HVAC residential applications.

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ACKNOWLEDGEMENTS

The authors thankfully acknowledge E.I. du Pont de Nemours and Company for supporting this work and for providing the refrigerant samples. The authors would also like to acknowledge the help from Auvi Biswas, a graduate research assistant at OSU, with the management and service of the psychrometric facility.